ANALYSIS OF AIR CONDITIONING EFFECTIVENESS VS. OUTDOOR CONDITIONS: TRADITIONAL BINS OR JOINT FREQUENCY BINS?

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ABSTRACT

There are a number of methods used to estimate the effectiveness of air conditioning equipment in handling loads. Full hourly computer simulations are probably the most accurate, but lack flexibility and are more cumbersome to use than more compact approaches. Alternately, some form of binned weather data has been used with load and performance estimation carried out for each of the bin weather conditions. The most common binning method puts weather into bins of dry bulb temperature with mean coincident wet bulb temperatures.

Mean coincident humidity terms lose the extreme humidity levels that commonly exist. This can lead one to assume that conditions will be held at all times, while in fact the humidity loads will not be met and discomfort, among other consequences, will result. Three-dimensional plots of the joint frequency results clearly illustrate problem areas.

A better procedure, it will be shown, is to use a joint frequency bin data set, which puts hours of occurrence into a matrix with dry bulb ranges on one axis and humidity ratio ranges on the second axis. This form of binning is easily accomplished if a utility like BinMaker™ is used to generate the binned data set.

INTRODUCTION

The factors which should be considered in the specification of heating and air conditioning equipment for buildings are first cost, operating cost and ability to maintain conditions in the face of variations in the load that the equipment must meet. The first cost is normally determined at what are known as the design conditions for the project. The operating cost is determined by a seasonal evaluation of the loads on the conditioned space and the cost of operation of the selected equipment to meet the aggregate loads. The ability to maintain comfort or set-point conditions is a function of the selected method of distributing the heating and cooling throughout a building and the ability to control the equipment to avoid wide excursions from the selected control points.

Evaluation of heating equipment is somewhat more straightforward than evaluation of air conditioning equipment in that meeting sensible loads is usually a sufficient criterion for acceptability, and conventional fossil-fuel-fired equipment has an efficiency which is relatively insensitive to the operating conditions. Historically, air conditioning equipment has been treated in the same way as heating equipment. However, the equipment must remove rather than add heat and humidity. Traditionally, calculations are done for both latent and sensible loads. Equipment is selected that can meet the total "thermal" requirement of these two loads at the specified design conditions which consist of a desired indoor temperature and humidity and a design point outdoor temperature and humidity.

The control used for the selected equipment is only a thermostat, which maintains comfortable temperature conditions but knows nothing about the status of the interior humidity. The control of humidity is a byproduct of the control of temperature. It occurs by virtue of the accompanying condensation of moisture from the air on the cooling coil that is doing the temperature control.

But, the loads are, in fact, quite different and missing either load is not a trivial problem. Missing the humidity set point can not be compensated for by a longer run time on the air conditioning system. The equipment simply stops running when the temperature set point is met leaving occupants in a humid climate feeling "clammy". The intuitive way to adjust is to lower the temperature set point. This does remove some more humidity by virtue of longer run times, but can lead to a "cold" and "clammy" feeling and the occupant does not know what to do about the discomfort. At certain times, when the sensible load drops to a low or zero value, but the humidity load remains, such as in the evenings when the sun is down and ambient temperatures drop, the occupant can be very uncomfortable.

In view of this problem, HVAC engineers are looking to newer approaches to meeting the loads which can accommodate these divergent demands on the system. Variable air flow systems or variable output systems attempt to vary the sensible heat ratio of a unit in response to measurement of both temperature and humidity in a space. Even more responsive to such needs are systems that integrate separate dehumidification and cooling functions, calling upon the appropriate one when the need exists. Such systems may consist of a desiccant-based unit, which may use a fuel like natural gas for its primary input, coupled with a conventional cold-coil unit largely for sensible cooling which generally, though not always, will use electrical power as its primary input.

The design point is selected such that it leaves a small fraction of the cooling season somewhat out of control. That is to say, the equipment may just meet the load at design conditions and be somewhat deficient at more severe conditions. Oversizing will compensate to some degree and the occupant theoretically should not feel discomfort. But, unlike in the heating season, where meeting a thermostat set point is the only criterion, in the cooling season, if conventional equipment is oversized to meet cooling loads beyond those at the design point, the poor match of humidity removal to humidity load is likely to leave the occupants uncomfortable anyway.

This paper presents the results of a simplified load analysis for distinctly different applications in a hot and humid climate. In such climates, the excursions in the humidity load compared to the sensible cooling load are more problematic. The load analysis will be extrapolated to a comfort analysis in order to demonstrate the consequences of poor humidity control using conventional air conditioning equipment.

BIN WEATHER OPTIONS

The analysis performed in this study was done utilizing a new method of analyzing weather data made possible by a new computer software application[1]. It allows the conversion of hourly TMY weather data into many more compact forms. One form is the familiar bin-type presentation, where hours of occurrence of close weather conditions are accumulated in groups or "bins". The analysis of loads and systems is done for the averaged values of temperature and humidity residing in each bin, rather than for every individual hour.

One common approach looks at monthly values of dry bulb temperatures aggregated into bins that span 5 degrees F. Along with each dry bulb temperature is the "mean coincident" wet bulb temperature or some other humidity parameter, such that loads can be analyzed for cooling using both temperature and humidity to do the calculations. Since humidity is an average value, this method necessarily loses some of the extremes.

In contrast to that approach, a joint frequency bin analysis can capture the extremes. Figure 1 is a sample of the joint frequency output that is possible from BinMaker™ for New Orleans, Louisiana, which is unquestionably a hot and humid climate. If one looks at New Orleans on an annual basis, the mean coincident humidities would be very misleading. For example, at the 81 F temperature level, the annual mean coincident humidity is about 113 gr/lb. However, out of the 409 hours in that temperature bin (80 - 82), there are over 250 hours above 115 gr/lb, indicating much higher dehumidification loads than one would expect from the mean value. The fact that there are hours at lower humidities doesn't help make one comfortable when the humidity is high. Comfort does not follow a mean value rule.

In the following sections, the joint frequency presentation of the weather data will be used to demonstrate the task that an air conditioner has in meeting the loads in extreme humidity conditions, and the resulting discomfort that occurs from not meeting them.

Figure 1: Sample Joint Frequency Distribution

Mid-pts	DB (F)	75	77	79	81	83	85	87	89	91	93	95	97
HR (gr/lb)	Bins	74 to 76	76 to 78	78 to 80	80 to 82	82 to 84	84 to 86	86 to 88	88 to 90	90 to 92	92 to 94	94 to 96	96 to 98
157.5	155 to 160	0	0	0	0	0	0	1	0	0	0	0	0
152.5	150 to 155	0	0	0	0	0	0	2	2	1	0	0	0
147.5	145 to 150	0	0	0	0	1	6	7	4	4	0	0	0
142.5	140 to 145	0	0	0	7	7	6	15	17	18	1	0	0
137.5	135 to 140	0	1	15	36	37	29	45	36	23	6	1	0
132.5	130 to 135	4	13	43	48	58	39	48	43	35	4	0	0
127.5	125 to 130	56	43	78	66	49	45	50	40	23	4	0	0
122.5	120 to 125	84	61	56	52	38	33	36	23	20	12	0	0
117.5	115 to 120	103	61	77	47	17	26	21	12	24	7	0	0
112.5	110 to 115	114	47	37	26	31	18	11	7	11	5	0	0
107.5	105 to 110	65	22	23	15	12	8	4	8	6	1	0	0
102.5	100 to 105	28	16	17	22	14	10	7	7	5	1	0	0
97.5	95 to 100	35	25	16	10	4	12	3	2	6	0	0	0
92.5	90 to 95	25	10	9	10	12	10	10	3	4	0	0	0
87.5	85 to 90	20	15	23	19	10	10	11	0	7	0	0	0
82.5	80 to 85	17	13	22	7	11	6	3	1	0	0	0	0
77.5	75 to 80	23	6	9	11	6	4	0	1	0	0	0	0
72.5	70 to 75	20	6	7	10	5	2	0	0	0	0	0	0
67.5	65 to 70	12	7	7	6	5	1	1	0	0	0	0	0

BUILDING LOADS ANALYSIS

In order to demonstrate some of the consequences of high humidity loads, a simplified building loads analysis method which is amenable to bin analysis has been composed. There are fundamentally six types of loads on a conditioned space, three each for both the sensible and latent components. They are:

- Ventilation load, which is the load brought in by fresh air and treated typically by mixing with return air and feeding the mixture to an air conditioning unit;

- Space transmission load, which is the load on the space itself due to differences in temperature and humidity between the indoor space and the outside conditions, like wall heat conduction and infiltration;

- Internal generation load, which is due to sources inside the conditioned space and independent of ambient conditions, such as generation of heat and perspiration from occupants, and lighting.

For any building, a design engineer would calculate these loads at design conditions and size his air conditioning equipment to meet this aggregate load, probably with some oversizing as a safeguard against inaccuracies in methods of calculation or wide excursions from design conditions. Having these loads allows one to calculate load coefficients that can be used for a bin or joint frequency analysis.

At design conditions, the loads can be defined as follows:

Q_{ssd} = space sensible load
Q_{svd} = ventilation sensible load
Q_{sgd} = internal sensible generation
Q_{hsd} = space humidity load
Q_{hvd} = ventilation humidity load
Q_{hgd} = internal humidity generation

Now, though the calculations to arrive at these loads can be complex, once they are available, assume that, where T_{des} and W_{des} are the designated outdoor design temperature and humidity and T_i and W_i are the desired indoor temperature and humidity:

$$Q_{ssd} = K_{ss} \text{ X } (T_{des} - T_i)$$
$$Q_{svd} = K_{sv} \text{ X } (T_{des} - T_i)$$
$$Q_{hsd} = K_{hs} \text{ X } (W_{des} - W_i)$$
$$Q_{hvd} = K_{hv} \text{ X } (W_{des} - W_i)$$

This now allows the simplified calculation of the variation in the loads with a set of weather conditions. As is typical of any bin or non-hour-by-hour analysis, the variations in occupancy and solar and wind conditions are neglected.

This type of calculation can be easily done by any engineer after completing the analysis of a building at design conditions. It has been used here to demonstrate the actual sensible heat ratios that the equipment must meet and to indicate how much the conditions will be missed by the equipment.

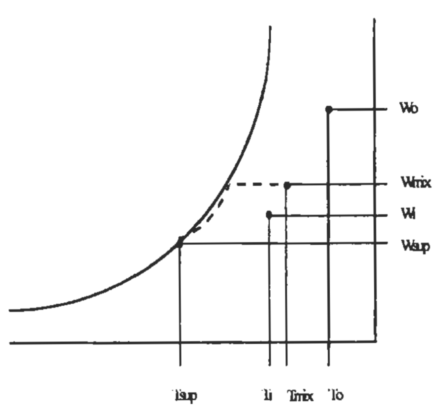
AIR CONDITIONING SYSTEM

An air conditioning system will generally be designed to meet the total calculated load, but controlled to meet the sensible load by thermostatic control. The objective of analyzing the air conditioning equipment here is to provide an indication of the failure of the equipment to meet loads in hot and humid climates. This paper presents a first look at these effects and it is anticipated that more comprehensive work will be done and reported at a later date.

Figure 2 is a psychrometric diagram showing how an idealized air conditioner works. The indoor and outdoor conditions are shown as T_i, W_i and T_o, Wo. For design calculations, T_o and W_o would be T_{des} and W_{des}. Fresh air is brought in as required usually by the occupancy. Fifteen to twenty cfm of fresh air per occupant is typical of ASHRAE Standard 62[2] recommendations and is used by most engineers for normal commercial applications today.

Figure 2

Psychrometric Representation of an Air Conditioner



The fresh air is mixed with return air and the mixture is fed to an air conditioning coil, either a chilled water coil or a direct expansion coil. The air off the coil is not typically saturated, but is close, so, for simplicity of analysis, the air is assumed to exit the air conditioner saturated, and that air is delivered to the conditioned space to handle the space loads. This is of course done by removing air at space conditions and delivering air at a temperature and humidity below space conditions, so that a net removal of heat and humidity occurs. The air at space conditions that is removed is either return air that goes back to the air conditioner to be cooled and dehumidified, or either exhaust or exfiltrated air. When fresh air is introduced to a space, as is required by building codes, space air must be either mechanically exhausted or it will naturally exfiltrate due to the pressurization effect of the fresh air.

For equilibrium to exist, the heat and humidity removal rates will be equal to the transmission and generation rates. If this is not the case, then the indoor conditions will change in the direction of establishing equilibrium. Inadequate dehumidification will cause the indoor humidity to rise until a moisture balance on the space is achieved. Excess dehumidification will cause the humidity to equilibrate at a lower level. Humidity mismatch is referred to here, because thermal match is generally provided by thermostatic control, and really only misses when extremely high loads occur (above the design loads).

The designer would assume a supply air temperature, T_{sup}, and a corresponding supply air humidity at or close to saturation, W_{sup}, and calculate an air flow to the space based upon meeting the space sensible load. The space sensible load is

$$Q_{sps} = Q_{ss} + Q_{sg}.$$

The required supply air flow, in lb/hr, is calculated by

$$F_{sup} = Q_{sps} / (0.24 \times (T_i - T_{sup})).$$

The supply flow is the sum of the return flow and the ventilation flow, so, if F_{vent} is the ventilation flow, then the return flow is

$$F_{ret} = F_{sup} - F_{vent}.$$

The air conditioner would typically be sized to meet the total load with some safety factor, but, neglecting safety factor, which doesn't help control humidity anyway, one can say that the capacity, C is

$$C = (Q_{ssd} + Q_{svd} + Q_{sgd} + Q_{hsd} + Q_{hvd} + Q_{hgd}).$$

Then the enthalpy change of the air across the air conditioner is

$$Dh_{ac} = C / F_{sup}.$$

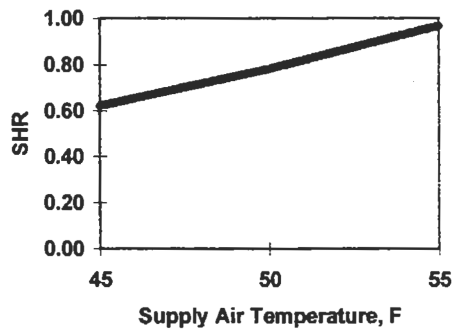
The analysis used herein makes use of the fact that the air conditioner will vary its operating temperature (supply air temperature) as it sees different entering air conditions. It will adjust itself and the space humidity will adjust itself until both the sensible and latent loads are met by the sensible and latent capacities of the air conditioner, and the space is in thermal and moisture equilibrium.

LOADS AND SENSIBLE HEAT RATIOS

The first item to consider is the nature of the load on the building. The critical aspect at design point (the conditions under which the air conditioning system is sized) is the space load. It should be possible to deliver air that will handle the design space load with the air conditioning unit running continuously. Since the unit delivers saturated or near-saturated air, the space sensible heat ratio[1], which is derived from the transmission and internal generation loads, should not be too low.

To illustrate this, Figure 3 describes the sensible heat ratio that is delivered to the space as a function of supply air temperature, assuming that the supply air is saturated. The space condition is 75F/50%rh. With a low supply air temperature of 45F, it is still only possible to drive the SHR down to about 0.62. At 55F, the capacity is all sensible. At a typical supply air temperature of 50F, the SHR is about 0.78.

Figure 3
Sensible Heat Ratio
Delivered to Space



Fortunately, at design conditions, the space sensible heat ratios for many common applications are fairly high. Offices that have a lot of sensible heat generation due to lighting and office equipment tend to be high, maybe greater than 0.9. (Other applications, like ice rinks that have internal cooling, can be low or even negative, and do not lend themselves to conventional air conditioning.)

However, at other, off-design conditions, this is not the case. A major factor is the ventilation air. It typically imparts a sensible heat ratio that is extremely low, possibly 0.2 or so, in mild, humid conditions. At such times, the air conditioner short cycles, because the space sensible load is low. Then, the fresh air will be brought into the space in the off cycle unconditioned and will become a space load on the air conditioner when the unit cycles on. This is best quantified by comparing the total load sensible heat ratio to the overall sensible heat ratio of the air conditioner when it is operating on the air which is a mixture of outdoor fresh air and indoor return air.

Figure 4 is a graph of the total sensible heat ratio on a 4000 sq.ft. office building in New Orleans as a function of outdoor air conditions. Presentation is limited to those conditions where

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[1] Sensible Heat Ratio (SHR) may be defined as the ratio of the sensible load to the total load or the ratio of the sensible cooling delivered to the space to the total cooling delivered to the space. The first term is sometimes called the sensible load ratio or the sensible heat factor. When the building is in equilibrium during air conditioning the load-related ratio is equal to the system-related ratio.

the ambient temperature and humidity are both higher than the indoor design conditions. The analysis described herein has been used to make this estimate. The design point data was taken directly from a sample calculation in the ASHRAE 1997 Handbook of Fundamentals[3].

As one might expect, even with the humid ambient in New Orleans, the total SHR is never much below about 0.5, which is not unreasonable for an air conditioning unit seeing a high humidity inlet condition derived from mixed humid ambient and normal return air. The SHR is as high as nearly 0.9 at warm, dry conditions. But this application has extremely high internal sensible gain derived from lighting (at about 5.8 W/sf).

Figure 4
Total Sensible Heat Ratio
New Orleans, ASHRAE small office

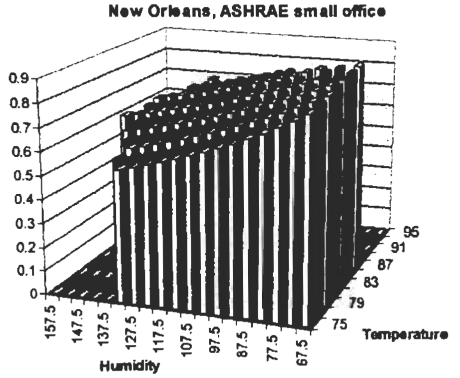
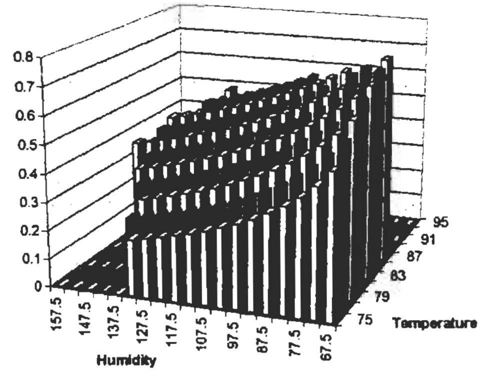


Figure 5 describes the SHR for the same building, but with the lighting load reduced to near zero and the occupancy doubled. This may be thought of as representing something like a movie theater or, possibly a night club. The difference in the two graphs of Total SHR as a function of ambient conditions (Figures 4 and 5) is striking. In Figure 5, the highest value of SHR which occurs is only around 0.7, and at cool and humid conditions it is as low as 0.2.

Figure 5
Total Sensible Heat Ratio
New Orleans, Theater



It is important to note that the shape of the surface describing the Total SHR is really not connected to the climate, so it is unnecessary to demonstrate this relationship for other climates. It is a characteristic of the application. It derives from the relationships or ratios among all the load contributions, the fixed latent and sensible loads and the variable latent and sensible loads.

An application more dominated by fixed loads, as in the case of the office building, will be flatter. If the dominant fixed loads are sensible, the average value will be high, and vice-versa if the dominant fixed loads are latent. When variable loads dominate, there is more slope to the surface, sloping downward toward high humidities if there are strong latent variable loads, like ventilation and infiltration, and sloping downward towards low temperatures if there are strong variable sensible loads. If both variations are strong, the surface will slope from high at low humidity and high temperature to low at high humidity and low temperature.

The climatic variation is accounted for by hours of occurrence of the conditions on the graph and will be discussed later.

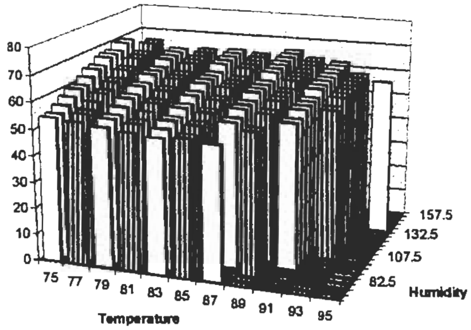
All of this may seem somewhat obvious now that it has been stated, but it says that various applications may be characterized by some type of surface SHR graph irrespective of their location. In fact, one may graph the surface for all realistic cooling conditions on the psychrometric chart and produce a family of characteristic shapes for a wide range of applications.

INDOOR CONDITIONS

As described, once the air conditioning equipment has been sized at a design point, it is possible to use a simplified analysis to estimate the degree to which the indoor humidity varies as ambient conditions vary. This was done for the office and theater applications for which the SHR's have been presented.

Figure 6 describes the equilibrium indoor humidity level for the office application in New Orleans. For the calculations in this study, the target indoor conditions were 75F and 65 grains/lb humidity or about 50% relative humidity. Temperature is the parameter that is assumed controlled by the thermostat, so the humidity excursions are taken to be an indicator of discomfort.

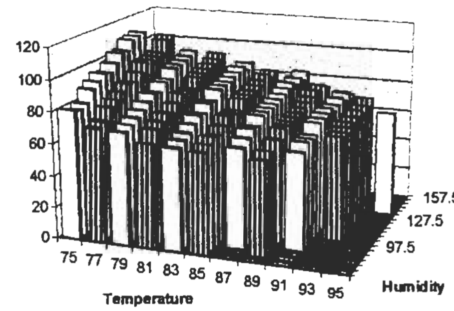
Figure 6
Equilibrium Indoor Humidity
New Orleans, ASHRAE small office



In the office case, even at low temperatures and high humidities, the dominance of internal sensible loads causes the air conditioner to run so long that it controls humidity quite well. In fact, room humidity peaks at about 71 grains/lb, and is generally driven below the desired indoor humidity by the long run-times of the air conditioner.

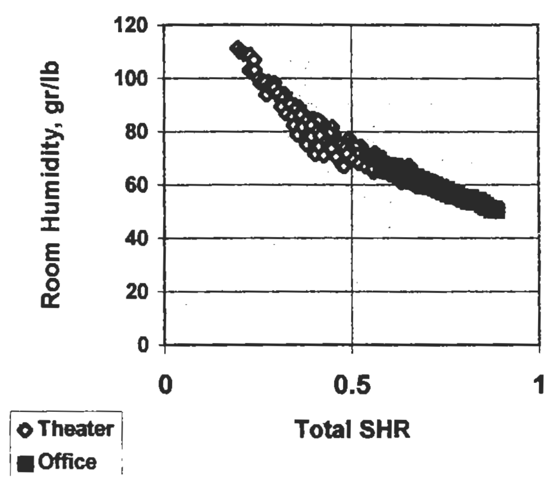
However, in the case of the movie theater, as shown in Figure 7, the excursions are quite large. At warm, dry conditions, there is no real problem, but the dominance of ventilation loads and internal people loads have caused the humidity to rise to over 110 grains/lb or to a relative humidity of about 85%. Anyone who has been in a crowded room in a humid climate knows that this is not unlikely.

Figure 7
Equilibrium Room Humidity
New Orleans, Theater



Again, the shape of the equilibrium humidity is more closely related to the application, but not uniquely as is the SHR relationship. Climate plays a role because, in different climates, the sizing of the air conditioning unit at design point would be different because the design point is different. The relationship of equilibrium humidity and Total SHR to the application, also causes them to relate to each other. A graph of this relationship is presented in Figure 8 for the theater and the office in New Orleans.

Figure 8:
Equilibrium Room Humidity vs, Total SHR
New Orleans, Office and Theater



DEGREE OF DISCOMFORT BY CLIMATE

While it has been shown that applications can be characterized as to their potential for creating uncomfortable conditions, by virtue of their inherent SHR characteristic, it is clear that the critical element that will affect the overall discomfort which occupants will feel in a particular application is the climate. This is to be expected, inasmuch as the highest level of discomfort would occur (for any application) when the ambient conditions are mild in temperature and humid.

Actually, it is likely that there is a correlation between relative humidity and discomfort. However, ambient humidities rarely would exceed 150 grains/lb. That would correspond to a relative humidity of 100% at 79F, but at 95F would only correspond to a relative humidity of 60%. That is in part why the discomfort seems not as severe at higher temperatures. If it were possible to experience high relative humidities at high temperatures, the situation would be even more different, but that would be unlikely.

To attain even 80% relative humidity at 95F would require a humidity ratio of over 200 grains/lb, which is not a realistic level. Humidity is generated in one way by a nearby body of water and the dew point would relate somewhat to the water temperature. For 200 grains/lb, the dew point is over 87F. Bodies of water are not usually that hot. The highest 0.4% humidity design point in the new 1997 ASHRAE Handbook of Fundamentals appears to be Ad Dahwah, Qatar, which is 184 grains/lb. The water in the Persian Gulf may be fairly hot, as indicated by the high design humidities for all the countries located there, with Al Zahran, Saudi Arabia, at 180 grains/lb.

To appreciate the effect of climate on discomfort, it is necessary to look at the hours of occurrence of weather conditions and determine how frequently discomfort, according to some measure, would occur. Figure 9 is a graph of hours of occurrence of bins of temperature and humidity for New Orleans when ambient conditions are above the designated indoor conditions of 75F and 65 gr/lb. One can see the tendency for hours to cluster at high humidities and moderate summer temperatures. Figure 10 shows the same relationship for Birmingham, Alabama, which is a climate somewhat like that of New Orleans, but not as extreme.

Figure 9
Hours of Occurrence
New Orleans, LA

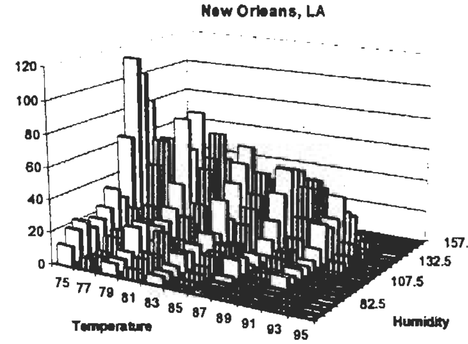
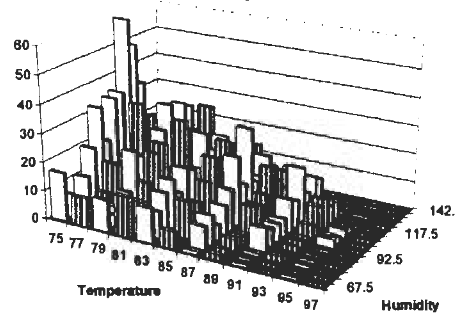


Figure 10
Hours of Occurrence
Birmingham, AL



In order to appreciate the degree of discomfort, one needs a standard against which to measure comfort. The ASHRAE 1997 Handbook of Fundamentals also can serve to help here (Figure 4, page 8.12). At an indoor temperature of 75F, ASHRAE would mark the top of the comfort zone at 91 grains/lb. Whether this is really comfortable or not is subjective. It corresponds to a wet bulb temperature of 68F, which is taken to be the upper boundary of the zone. It also indicates a relative humidity of 70% which most people would probably not consider comfortable, and is on the edge of the mildew region. Even at 80 grains/lb, the relative humidity is 60%.

Table 1 shows the number of hours that the indoor humidity would exceed various thresholds for both New Orleans and Birmingham for the theater application, out of the total number of hours in the range shown in the graphs of "hours of occurrence". In New Orleans, out of a possible 3075 hours, the indoor humidity will exceed 80 grains/lb for 2130 hours. In Birmingham, out of a possible

2043 hours, there will be 1070 hours when the humidity will exceed 80 grains/lb. In New Orleans, there are even 375 hours when the indoor humidity will exceed 100 grains/lb. It is interesting to note further that there are many hours when the humidity is high, but the dry bulb temperature is below the indoor set point. Those hours would create high indoor humidities, but are not shown here, because they are assumed not to be in the air-conditioning season.

Table 1. Hours Exceeding Humidity

Hours	New Orleans	Birmingham
Total	3075	2043
Over 80	2130	1070
Over 90	1237	564
Over 100	375	196

Another difference between the theater and the office is the equipment which would be selected for meeting the load at the design point. For the office, in New Orleans, the application requires 23% ventilation air and operates at 285 cfm/ton. This sizing is not unreasonable for conventional air conditioners. On the other hand, for the theater, the low internal sensible load and the higher ventilation rates result in 76% ventilation air and an operation at 155 cfm/ton. The latter sizing is especially problematic because of the high ventilation component. When ambient conditions are milder, the installed capacity can create extremely low coil temperatures, even to the point of freezing on the coil, unless capacity control is employed. So, not only does conventional equipment have a hard time controlling comfort in the face of low total building sensible heat ratios, but it requires extra control measures that add significantly to the expense of the equipment.

CONCLUSION

It has been shown that applications can be simply categorized by a trend on an SHR versus ambient conditions presentation. Some applications, which promote high SHR levels lend themselves to easy control of comfort over a wide range of operating conditions, using conventional equipment. Others, for which SHR levels tend to be lower, do not.

Of particular concern are applications that require a large portion of ventilation air. For a movie theater-like situation, indoor equilibrium humidity levels can be shown to exceed 100 grains/lb for a significant number of hours in a Gulf Coast climate like that of New Orleans, if conventional approaches to air conditioning are used. Of additional concern are the unconventional control and sizing associated with high ventilation fraction air conditioning systems. More applications need to be characterized to get a better picture of the trends.

In order to alleviate the problems that have been identified, a method of efficiently controlling temperature and humidity independently should be employed. This study has not reached the point of a comparative evaluation of the various solutions that could be considered.

It should be borne in mind that, although this analysis was done for complete single zone buildings, the problems that have been illustrated can occur in individual zones of a larger building if they are individually controlled and ventilated. For example, a hospital surgical room which use 100% fresh air to clear bacteria would be similar, even if its cooling is from a chilled water coil that is fed by a chiller that services other parts of the hospital as well.

One can now obviously see the need for considering the milder and more humid conditions that occur so often in the SouthEast United States. One way to be sure of handling these situations without complete joint frequency analysis is to utilize the new ASHRAE weather tables in the 1997 Handbook of Fundamentals, which list design humidity conditions with mean coincident dry bulb temperatures. This would convey to the designer whether or not he will have a serious problem with humidity control if he designs only for the conventional design conditions.

One aspect of the problem that has not been discussed is conditions where sensible loads are such that air conditioning never runs on thermostatic control but outdoor humidities are high. This analysis does not treat that situation.

For such a situation, without independent humidity control, one could expect indoor humidities to be the same as or similar to outdoor humidities, which can be very uncomfortable.

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